MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 1, 2019/2020

EME3276 – MECHANICAL DESIGN II

(ME)

25 OCTOBER 2019
9.00 am – 11.00 am
(2 Hours)

INSTRUCTIONS TO STUDENT

1. This question paper consists of 12 pages including cover page.
2. Attempt ALL questions.
3. Please write all your answers in the answer booklet provided. Show all relevant steps to obtain maximum marks.
4. You may use the appendices provided in the booklet.

Question 1

The worm shaft depicted in **Figure Q1** part (a) runs at a speed of 600 rev/min and transmits a power of 900 W. A static force analysis gave the results shown in part (b) of the figure. It is suggested to mount an angular-contact ball bearing at A to take the 2500 N thrust load, and a straight roller bearing at B to take only radial load. Assuming light impact loading with an application factor of 1.3, a desired life of 18 kh, a combined reliability of 0.99, and both bearings have inner ring rotation,

a) Select a straight roller bearing at B. **[8 marks]**

b) Select an angular-contact ball bearing at A. (*Hint: This selection procedure requires iterative procedure. Take $X_2 = 0.56$ and $Y_2 = 1.85$ as the first trial*) **[17 marks]**

[Note: The bearing rating life, $L_R = 10^6$ revolution and Weibull parameters for ball and straight roller bearings, $x_0 = 0.02$, $(\theta - x_0) = 4.439$ and $b = 1.483$.]

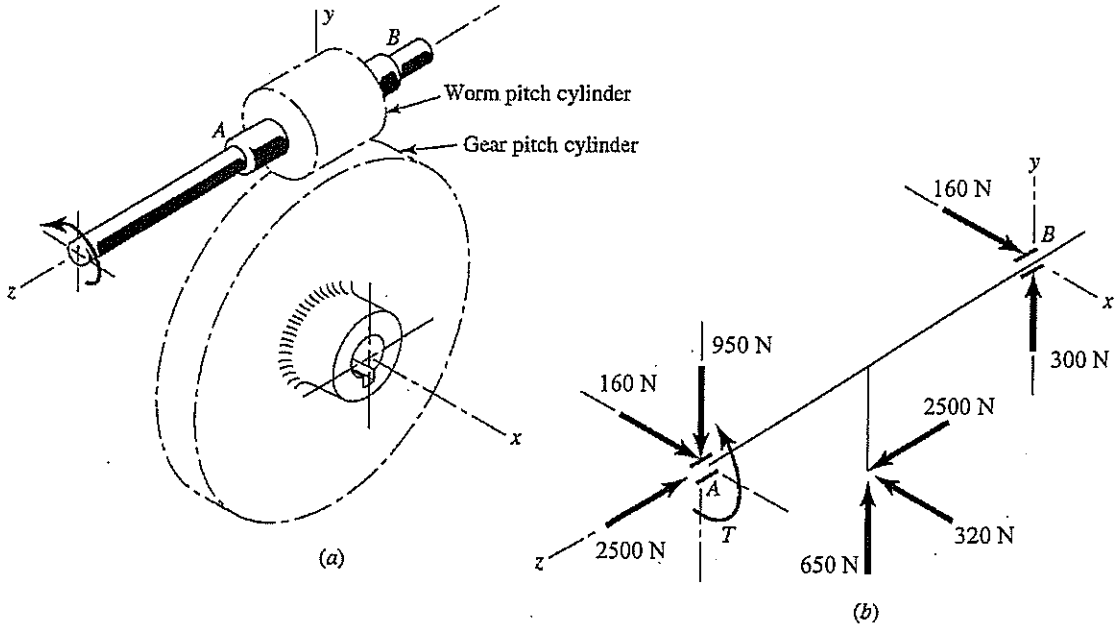

Figure Q1

Continued...

Question 2

Design a suitable helical compression spring for a clamping fixture as shown in **Figure Q2**. The fixture is to hold a workpiece of 32.5 mm thick. An A227 Hard Drawn wire spring is required to drive the clamp upward when removing the workpiece with a starting force of 55 N. The clamp screw has an M12 × 1.75 thread. Allow a diametral clearance of 1.75 mm between it and the uncompressed spring. It is further specified that the free length of the spring should be $L_0 \leq 52$ mm and the safety factor when closed solid should be $n_s \geq 1.2$. Use the wire diameter, $d = 2$ mm in this design and assume squared and ground ends.

[25 marks]

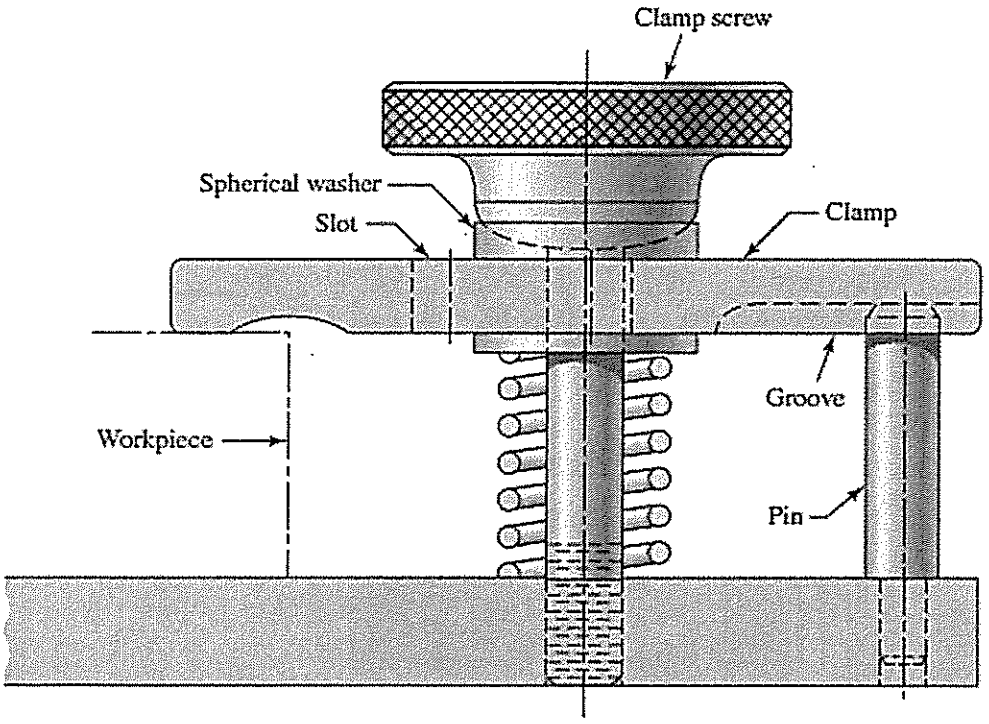

Figure Q2

Continued...

Question 3

A long shoe drum clutch, as shown in **Figure Q3,** is used to produce a friction torque of 85 Nm to stop a drum from counter-clockwise rotation at 120 rpm. It has rim radius, $r = 100$ mm and is actuated by force W. Given the following parameters:

$$C = 200 \text{ mm}; L = 380 \text{ mm}; \theta_1 = 30°; \theta_2 = 150°$$

The clutch lining is a moulded compound and has a coefficient of friction of 0.25 with mating surface. It has a maximum lining pressure capacity up to 0.52 MPa.

c) Estimate the minimum required brake pad face width, w, to the nearest mm. **[3 marks]**

d) With the determined brake pad face width in **part a)**, examine if the maximum pressure at $\theta = 90°$ is within the lining pressure capacity. **[3 marks]**

e) Draw the appropriate free body diagram(s) to show the relationship between the actuating force W and the resulting reaction at the pivot. Determine both the actuating force W and the reaction at the pivot. **[16 marks]**

f) Suggest two ways to minimize the actuating force for the required friction torque. **[3 marks]**

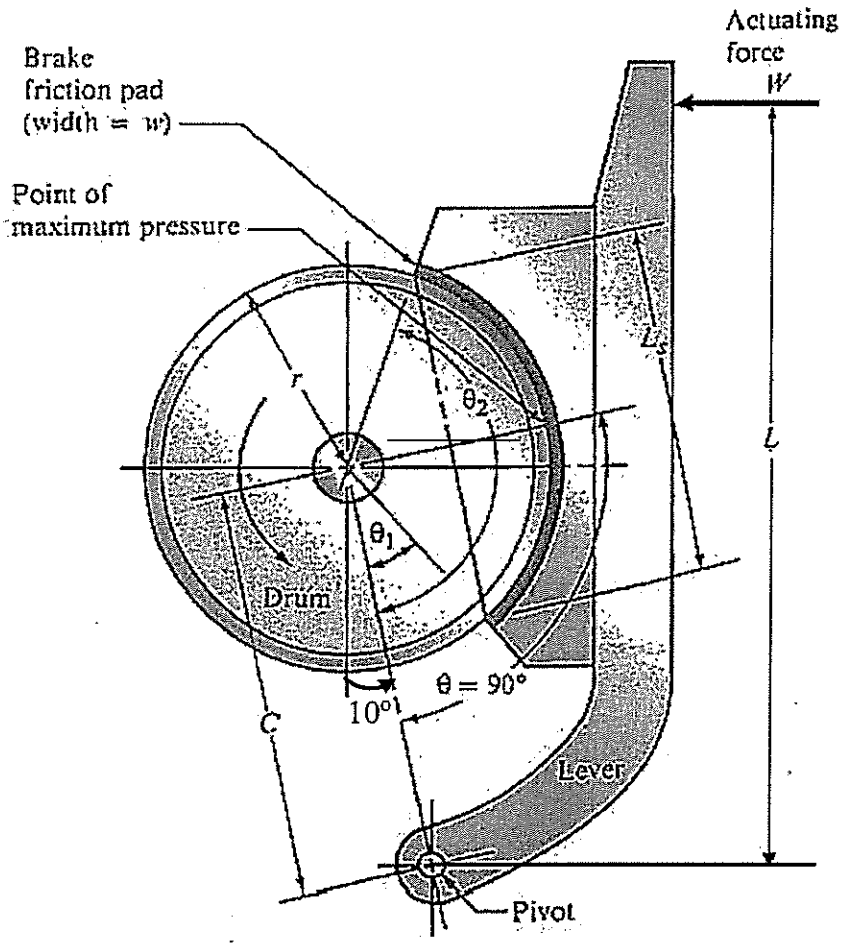

Figure Q3

Continued...

Question 4

For the band brake shown in **Figure Q4**, the following parameters are given:

Drum diameter, $d = 300$ mm;
Brake lining maximum pressure, $p_{max} = 1.5$ MPa;
Brake pad material friction coefficient, $\mu = 0.3$;
Brake pad face width, $b = 50$ mm.

The band brake is to designed to achieve a minimum braking torque of 1000 Nm, with the maximum actuating force of 2 kN.

a) Analyse the brake system to determine if the required braking torque can be achieved. **[18 marks]**

b) Draw the free body diagram(s) necessary to compute for the forces acting at hinge *O*. In the diagram(s), indicate the forces and torques in equilibrium clearly. **[4 marks]**

c) Suggest two dimensional changes that will increase the braking torque while maintaining the actuating force. **[3 marks]**

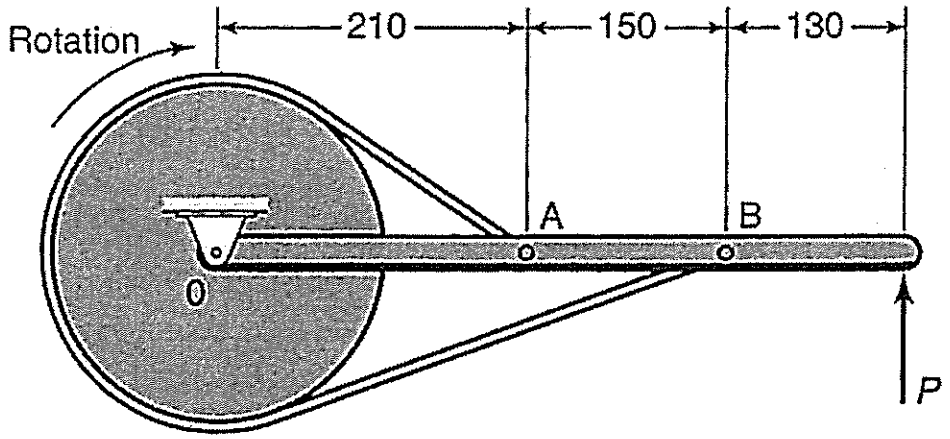

Figure Q4
[All dimensions in mm]

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EQUATIONS

BEARING

$$x_D = \frac{L_D}{L_R} = \frac{60 L_D n_D}{60 L_R n_R}$$

where $L_R = L_{10} = 10^6$ revolutions

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)[\ln(1/R_D)]^{1/b}} \right]^{1/a}$$

$a = 3$ for ball bearing and $a = 10/3$ for roller bearing;
$b = 1.483$; $x_o = 0.02$; $(\theta - x_o) = 4.439$ unless otherwise mentioned

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a}$$

for $R_D \geq 0.90$

$$F_e = X_i V F_r + Y_i F_a$$

$i = 1$ when $F_a/(VF_r) \leq e$ and $i = 2$ when $F_a/(VF_r) \geq e$.

$V = 1$ for inner ring rotation;
$V = 1.2$ for outer ring rotation

$$R_D \approx 1 - \left\{ \frac{x_D \left(\frac{a_f F_D}{C_{10}} \right)^a - x_0}{\theta - x_0} \right\}^b$$

for $R_D \geq 0.90$

SPRING

$$C = D/d$$

$$\tau_s = K_B \frac{8FD}{\pi d^3}$$

$$k = \frac{F}{y} = \frac{d^4 G}{8D^3 N_a}$$

$$S_{ut} = \frac{A}{d^m}$$

$$K_B = \frac{4C + 2}{4C - 3}$$

$$n_s = \frac{S_{sy}}{\tau_s}$$

$$L_0 < \frac{\pi D}{\alpha} \left[\frac{2(E - G)}{2G + E} \right]^{1/2}$$

$$L_0 < 2.63 \frac{D}{\alpha}$$ for steels

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CLUTCHES AND BRAKES

External Contracting Rim Clutches and Brakes

$$M_f = \frac{fp_a br}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta(r - a\cos\theta)d\theta = \frac{fp_a br}{\sin\theta_a}\left(-r\cos\theta - \frac{a}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$M_N = \frac{p_a bra}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \frac{p_a bra}{\sin\theta_a}\left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

$$T = \frac{fp_a br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta d\theta = \frac{fp_a br^2}{\sin\theta_a}(\cos\theta_1 - \cos\theta_2)$$

$$A = \int_{\theta_1}^{\theta_2}\sin\theta\cos\theta d\theta = \left(\frac{1}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$B = \int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

Counter-clockwise rotation: (Right Shoe)

Self-energizing ($M_N > 0$, $M_f > 0$):

$$F = \frac{M_N - M_f}{c}$$

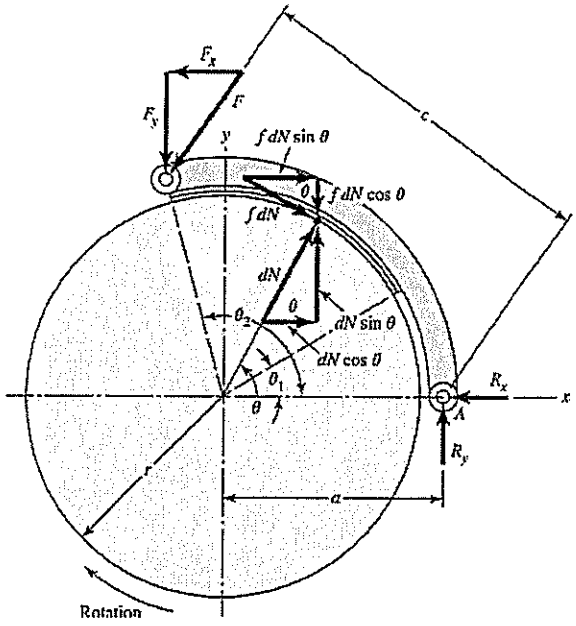
$$R_x = \frac{p_a br}{\sin\theta_a}(A - fB) - F_x$$

$$R_y = \frac{p_a br}{\sin\theta_a}(-fA - B) + F_y$$

Clockwise rotation: (Right Shoe)

Non-self-energizing ($M_N > 0$, $M_f > 0$):

$$F = \frac{M_N + M_f}{c}$$



Band Type Clutches and Brakes

$$\frac{P_1}{P_2} = e^{f\phi}$$

$$P_1 = p_{\max} br$$

$$T = (P_1 - P_2)\frac{D}{2}$$

Continued.....

TABLES

BEARING

Table 1: Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 2: Dimensions and Load Ratings for Single-Row 02-Series Deep Groove and Angular-Contact Ball Bearings

			Fillet	Shoulder		Load Ratings, kN			
Bore,	OD,	Width,	Radius,	Diameter, mm		Deep Groove		Angular Contact	
mm	mm	mm	mm	d_S	d_H	C_{10}	C_0	C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

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Table 3: Dimensions and Load Ratings for Cylindrical Roller Bearings

	02-Series				03-Series			
Bore,	OD,	Width,	Load Rating, kN		OD,	Width,	Load Rating, kN	
mm	mm	mm	C_{10}	C_0	mm	mm	C_{10}	C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

Table 4: Formulas for the Dimensional Characteristics of Compression-Springs

Term	Type of Spring Ends			
	Plain	Plain and Ground	Squared or Closed	Squared and Ground
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

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Table 5: End-Condition Constants α for Helical Compression Springs

End Condition	Constant α
Spring supported between flat parallel surfaces (fixed ends)	0.5
One end supported by flat surface perpendicular to spring axis (fixed); other end pivoted (hinged)	0.707
Both ends pivoted (hinged)	1
One end clamped; other end free	2

*Ends supported by flat surfaces must be squared and ground.

Table 6: Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of Wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire†	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire‡	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire§	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‖	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire#	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire**	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

Table 7: Maximum Allowable Torsional Stresses for Helical Spring in Static Applications

Material	Maximum Percent of Tensile Strength Before Set Removed (includes K_W or K_B)	Maximum Percent of Tensile Strength After Set Removed (includes K_s)
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

Continued.....

Table 8: Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Torsion	Diameter d, in	E Mpsi	E GPa	G Mpsi	G GPa
Music wire A228	65–75	45–60	<0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			>0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	<0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			>0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Chrome-vanadium A231	88–93	65–75		29.5	203.4	11.2	77.2
A232	88–93			29.5	203.4	11.2	77.2
Chrome-silicon A401	85–93	65–75		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3
Phosphor-bronze B159	75–80	45–50		15	103.4	6	41.4
Beryllium-copper B197	70	50		17	117.2	6.5	44.8
	75	50–55		19	131	7.3	50.3
Inconel alloy X-750	65–70	40–45		31	213.7	11.2	77.2

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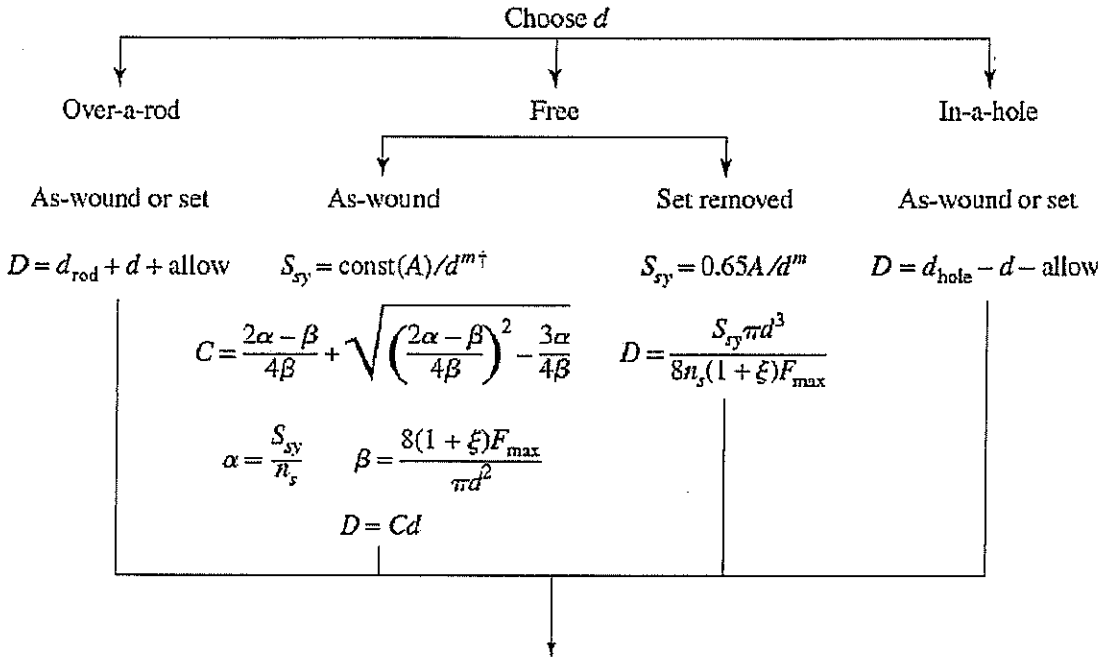
FIGURES

SPRING

Figure 1: Design Flowchart for Static Loading



$$C = D/d$$

$$K_B = (4C + 2)/(4C - 3)$$

$$\tau_s = K_B 8(1 + \xi)F_{max}D/(\pi d^3)$$

$$n_s = S_{sy}/\tau_s$$

$$\text{OD} = D + d$$

$$\text{ID} = D - d$$

$$N_a = Gd^4 y_{max}/(8D^3 F_{max})$$

$$N_t$$

$$L_s$$

$$L_0$$

$$(L_0)_{cr} = 2.63D/\alpha$$

$$\text{fom} = -(\text{rel. cost})\gamma\pi^2 d^2 N_t D/4$$

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